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Thermodynamic analysis of tri-generation systems taking into account refrigeration, heating and electricity load demands

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ABSTRACT

Tri-generation is a novel application of energy technologies which simultaneously produces heat, refrigeration and electricity. An expression for the calculation of the thermodynamic performance of a generic tri-generation scheme is presented. A brief first-law analysis involving an energy conversion ratio and newly defined heating-to-cooling and electric-to-cooling load ratios to usual system component thermodynamic parameters (such as coefficient of performance or prime mover thermal efficiency) was carried out. To illustrate the usefulness of the criterion, a tri-generation pilot plant set up in an office building was studied.

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1. Introduction

Tri-generation, the simultaneous production of electricity, heat and refrigeration from a primary source of energy, such as natural gas or bio-fuel, is a natural extension of co-generation. From a strictly thermodynamic viewpoint, a tri-generation system is simply a traditional combined heat and power (CHP) system plus an absorption and/or a vapor compression chiller (CCHP-combined cooling heating and power). However, the advantages of tri-generation, such as primary energy savings and greater overall efficiency have, in recent years, attracted authors, researchers and the construction community [1–11]. Tri-generation is clearly of importance in connection with pollution control. According to Meunier [12], who studied the impact of co- and tri-generation on the environment, CO_2 emissions could be reduced by at least 40% if sorption heat pump technology were to be developed. Moreover, tri-generation plants are economically viable in situations where electric energy is scarce and/or costly. Despite the attractiveness of tri-generation as an energy-integrated scheme, new energy demands arise, implying further operational constraints not present in co-generation.

In most tri-generation systems heat, refrigeration and electricity are produced by a combination of an absorption chiller and a Diesel or gas turbine generator. While the choice of a Diesel engine or gas turbine as a prime mover is dictated by a thermodynamic cost–benefit relation, the possibility of transforming low grade heat from CHP units into cold has made absorption chillers an almost indispensable component of most commercial applications. Absorption chillers offer good partial load performance, low maintenance and high availability and this may account for the fact that most research in the area has concentrated on tri-generation with absorption chillers. Some investigations [13,14] have focused on the food industry and supermarkets where items must be kept chilled or frozen in refrigerated cabinets throughout the year, subject to appreciable seasonal variations. These investigations highlighted the possible gains to be had from this new energy technology. However, less attention seems to be paid to the more important problem of the assessment of thermodynamic performance. Colonna and Gabrielli [15] studied tri-generation plants consisting of gas turbines and internal combustion engines driving ammonia-water absorption chillers. They defined an index of electric equivalent efficiency without, however, taking into account the various modes of energy encountered in tri-generation. Nevertheless, they confirmed the expected superiority of internal combustion engines over gas turbines by comparing the cold produced by tri-generation with that generated by a conventional compression chiller plant. In an interesting paper, Pépin Magloire and Bolle [16] linked the performance of tri-generation to that of co-generation, a century-old proven technology. Their "index d'économie d'énergie primaire" (index of primary energy economy) provides a useful measurement of performance. Later, Minciuc et al. [17] have shed some light on the calculation of the energy conversion efficiency of a tri-generation configuration by defining "energy

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Nomenclature

COP_c	cooling coefficient of performance
COP_h	heating coefficient of performance
ECR	energy conversion ratio
$\dot{E}_{load}$	electricity load [kW]
$\dot{H}$	energy rate equivalent of fuel consumption [kW]
$\dot{Q}$	heat transfer rate [kW]
$\dot{Q}_{cd}^{i}$	rate of heat transfer from condenser of chiller i [kW]
$\dot{Q}_{cooling}$	cooling load [kW]
$\dot{Q}_{heating}$	heating load [kW]
$\dot{Q}_{recovery}$	rate of total heat recovery [kW]
R_{CE}	cooling-to-electricity load ratio
R_{CE}^{*}	limit cooling-to-electricity load ratio
R_{FE}	energy rate equivalent of fuel consumption to electricity load ratio
R_{FE}^{*}	limit fuel consumption energy rate equivalent to electricity load ratio
R_{HE}	heating-to-electricity load ratio
R_{HE}^{*}	limit heating-to-electricity load ratio
$\dot{W}$	shaft power [kW]
$\dot{W}_{cp}^{i}$	shaft power to drive compressor of chiller i [kW]
x	fraction of the heat recovered from the exhaust and water cooling that goes to the building heat exchanger

Greek letters

α	fraction of heat engine energy-equivalent fuel consumption rate
Γ_{he}	overall heat recovery efficiency from heat engine
ε	heat recovery efficiency of heat exchanger
λ	cooling load distribution ratio
Σ_c	overall cooling efficiency factor
Σ_h	overall heat pump heating efficiency factor
η	efficiency

Subscripts

bo	boiler heat exchanger
bx	building heat exchanger
c	cooling
cd	condenser
co	heat engine coolant
cp	compressor
eg	electric generator
es	heat engine shaft
ev	evaporator
ex	heat engine exhaust
h	heating
he	heat engine
pb	peak boiler or pressurized auxiliary boiler
st	steam turbine
total	total

Superscripts

ac	absorption chiller
ec	electrically driven vapor compression chiller
sc	steam turbine driven vapor compression chiller

production" and "energy structure" indices to assess thermodynamic performance.

The aim of the present work is to derive an expression for the overall energy conversion efficiency of typical tri-generation systems in terms of two non-dimensional parameters, the heating-to-electric and the cooling-to-electric load ratios and in terms of the efficiencies of the system components. First, a generic tri-generation system is studied, followed by the application of the criterion to two practical cases.

2. Generic tri-generation system

2.1. System description

Fig. 1 shows, schematically, the system in question. A heat engine, which may be a gas turbine, a reciprocating internal combustion engine or even a Stirling engine, drives an electric generator. Electricity from the generator is used to power the vapor compression chiller which, apart from producing cold, rejects heat from the condenser. The electricity demand is met by the surplus of electricity production from the generator whereas the heat demand is met by recovering heat from the heat pump and from the heat engine exhaust gases and, in the case of reciprocating engines, from the coolant system as well. A heat recovery boiler extracts rejected heat from the engine exhaust gases, supplying steam to a steam turbine which, in turn, drives a vapor compressor chiller. Heat is recovered from the condensers of the three heat pumps. Such a generic system will hopefully be of use in the assessment of real systems.

The heat engine is characterized by the fuel energy distribution, to the shaft, exhaust and coolant, α_{es}, α_{ex} and α_{co}, respectively. The sum of α_{es}, α_{ex} and α_{co} is not necessarily equal to 1, as there are heat losses that are not recovered. In this generic case, it is assumed that the temperature at which heat is recovered from the engine coolant is high enough to drive the absorption chiller. In the case of a gas turbine, heat is extracted from the exhaust only, as there is no coolant fraction, and the two values of α take a different representation, corresponding to the fuel energy fractions that go to the heat recovery boiler and to the absorption chiller. The efficiency of the heat recovery boiler is ε_{ex}. The corresponding Rankine cycle, required for the conversion of steam from the heat recovery boiler to the steam turbine shaft work, is not represented in Fig. 1, its overall efficiency being characterized by η_{st}. Each heat pump, the electrically driven vapor compression, the steam turbine driven vapor compression and the absorption chillers, are each represented by a pair of COPs, for heating and cooling. Again, in contrast to the case of an ideal heat pump, the difference between the two COPs, heating and cooling, may not be equal to unity, due to the heat losses and gains.

The diagram of Fig. 1 shows that the tri-generation scheme runs entirely on a single energy source (denoted by "fuel") and has three so-called energy products, namely heat, refrigeration and electricity.

2.2. Thermodynamic model

The following assumptions are made:

(a) The vapor compression and absorption chillers are sized to supply the entire cooling load;
(b) the heat engine/electric generator compound is designed to provide sufficient power to meet the electricity demand and to drive the heat pump compressor;
(c) the temperature at which rejected heat is recovered is sufficiently high to meet the heat load demand and to drive the absorption chiller.

These assumptions deserve some comments. A peak boiler is indispensable in meeting the heat demand as all (energy) demands are hardly ever met at the same time (e.g., Teopa Calva et al. [18]). Furthermore, only a high temperature heat rejection is

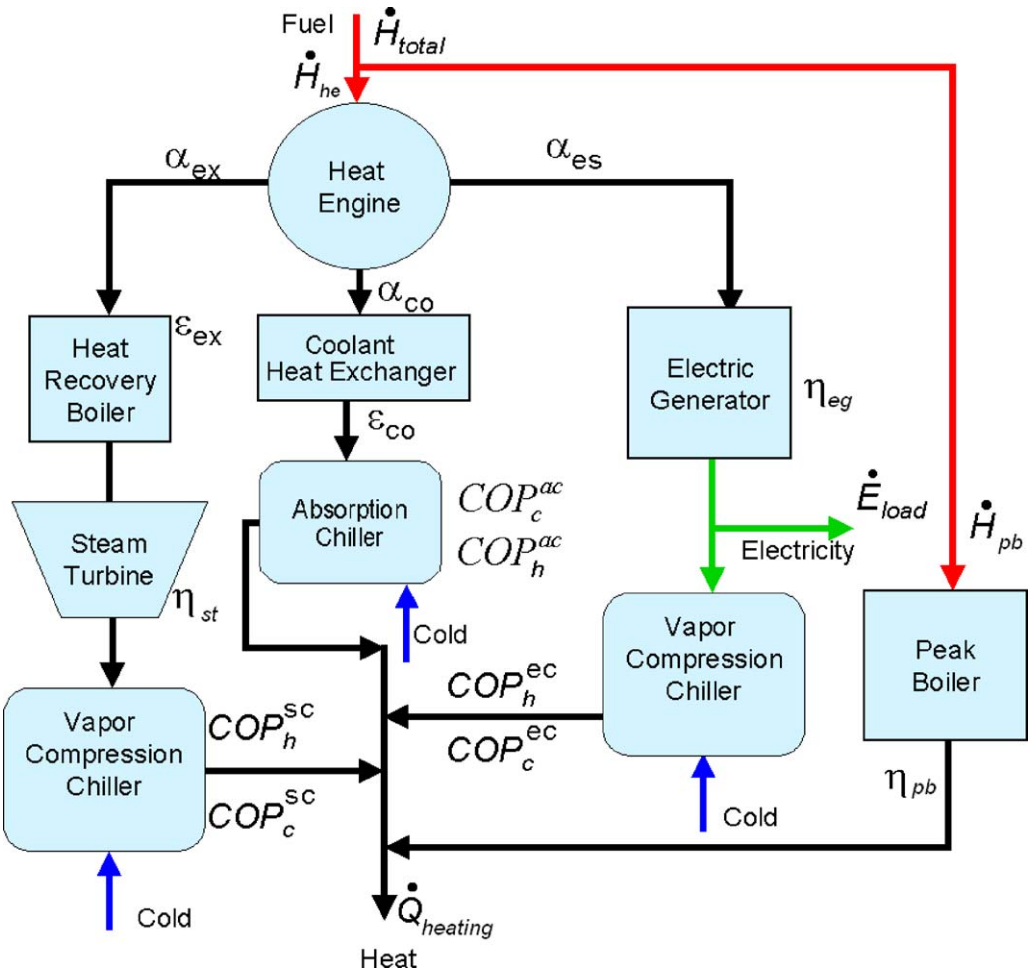

Fig. 1. Schematic diagram of a generic tri-generation system.

physically meaningful, for otherwise the system would lose its "tri-generation" character.

To model the tri-generation system, a set of parameters was taken from studies of heat recovery from a Diesel engine driven water-to-water heat pump by Parise and Cartwright [19] and of a refrigeration plant by Herbas et al. [20].

The energy conversion ratio, *ECR*, is defined as the total energy delivery (heat, electricity and refrigeration) divided by energy input, i.e., the fuel burned in the engine and in the peak boiler:

$$ECR = \frac{\dot{Q}_{cooling} + \dot{Q}_{heating} + \dot{E}_{load}}{\dot{H}_{total}} \tag{1}$$

Two other dimensionless ratios, R_{HE} and R_{CE}, compare the magnitudes of the heating, cooling and electricity loads:

$$R_{HE} = \frac{\dot{Q}_{heating}}{\dot{E}_{load}} \tag{2}$$

$$R_{CE} = \frac{\dot{Q}_{cooling}}{\dot{E}_{load}} \tag{3}$$

Using the fact that total fuel consumption is distributed between the peak boiler and the engine, both burning the same type of fuel:

$$\dot{H}_{total} = \dot{H}_{he} + \dot{H}_{pb} \tag{4}$$

one has:

$$ECR = \frac{\dot{Q}_{cooling} + \dot{Q}_{heating} + \dot{E}_{load}}{\dot{H}_{he} + \dot{H}_{pb}} \tag{5}$$

and, from Eqs. (2) and (3):

$$ECR = \frac{R_{CE} + R_{HE} + 1}{(\dot{H}_{he} + \dot{H}_{pb})/\dot{E}_{load}} \tag{6}$$

The heating load is met by the total heat recovered and by the peak boiler:

$$\dot{Q}_{heating} = \dot{Q}_{recovery} + \dot{Q}_{pb} \tag{7}$$

Since heat is recovered from the condensers of the absorption and vapor compression heat pumps, Fig. 1, the total heat recovery rate is given by:

$$\dot{Q}_{recovery} = \dot{Q}_{cd}^{ec} + \dot{Q}_{cd}^{sc} + \dot{Q}_{cd}^{ac} \tag{8}$$

For the peak boiler, of efficiency η_{pb}:

$$\dot{Q}_{pb} = \eta_{pb}\dot{H}_{pb} \tag{9}$$

Rejected heat is recovered from the engine coolant (to drive the absorption chiller) and exhaust gases (to steam turbine) at rates that depend on the energy balance of the heat engine, characterized by α_{co}, α_{ex} and α_{es}, and on the heat recovery efficiencies of both engine exhaust boiler and coolant heat exchanger, ε_{ex} and ε_{co}, so that:

$$\alpha_{es} = \frac{\dot{W}_{he}}{\dot{H}_{he}} \tag{10}$$

$$\varepsilon_{co}\alpha_{co} = \frac{\dot{Q}_{co}}{\dot{H}_{he}} \tag{11}$$

$$\varepsilon_{ex}\alpha_{ex} = \frac{\dot{Q}_{ex}}{\dot{H}_{he}} \tag{12}$$

Electric power is supplied to one of the vapor compression chillers, and both cooling and heating power are obtained from the evaporator and condenser, respectively. The cooling and heating coefficients of performance describe the energy balance in the electrically driven vapor compression heat pump cycle:

$$COP_c^{ec} = \frac{\dot{Q}_{ev}^{ec}}{\dot{W}_{cp}^{ec}} \tag{13}$$

$$COP_h^{ec} = \frac{\dot{Q}_{cd}^{ec}}{\dot{W}_{cp}^{ec}} \tag{14}$$

Taking into account the fraction of fuel energy that goes to power shaft in the heat engine, and the efficiency of the electric generator, the compressor electrical power consumption is given by:

$$\dot{E}_{load} + \dot{W}_{cp}^{ec} = \eta_{eg}\alpha_{es}\dot{H}_{he} \tag{15}$$

From Eqs. (13)–(15), the cooling capacity and condenser power output of the electrically driven chiller are, respectively:

$$\dot{Q}_{ev}^{ec} = COP_c^{ec}\dot{W}_{cp}^{ec} = COP_c^{ec}(\eta_{eg}\alpha_{es}\dot{H}_{he} - \dot{E}_{load}) \tag{16}$$

$$\dot{Q}_{cd}^{ec} = COP_h^{ec}\dot{W}_{cp}^{ec} = COP_h^{ec}(\eta_{eg}\alpha_{es}\dot{H}_{he} - \dot{E}_{load}) \tag{17}$$

For the steam turbine driven vapor compression chiller, similar equations can be written:

$$\dot{Q}_{ev}^{sc} = COP_c^{sc}\eta_{st}\varepsilon_{ex}\alpha_{ex}\dot{H}_{he} \tag{18}$$

$$\dot{Q}_{cd}^{sc} = COP_h^{sc}\eta_{st}\varepsilon_{ex}\alpha_{ex}\dot{H}_{he} \tag{19}$$

For the heat-driven absorption chiller, the equations become:

$$\dot{Q}_{ev}^{ac} = COP_c^{ac}\varepsilon_{co}\alpha_{co}\dot{H}_{he} \tag{20}$$

$$Q_{cd}^{ac} = COP_h^{ac}\varepsilon_{co}\alpha_{co}\dot{H}_{he} \tag{21}$$

The cooling load is provided by the three chillers:

$$\dot{Q}_{cooling} = \dot{Q}_{ev}^{ec} + \dot{Q}_{ev}^{sc} + \dot{Q}_{ev}^{ac} \tag{22}$$

Summing the three equations for $\dot{Q}_{ev}^{ec}$, $\dot{Q}_{ev}^{sc}$ and $\dot{Q}_{ev}^{ac}$, (16), (18) and (20), one has:

$$\dot{Q}_{cooling} = \Sigma_c\dot{H}_{he} - COP_c^{ec}\dot{E}_{load} \tag{23}$$

or

$$\frac{\dot{H}_{he}}{\dot{E}_{load}} = \frac{R_{CE} + COP_c^{ec}}{\Sigma_c} \tag{24}$$

where an overall cooling efficiency factor, Σ_c, can be defined as:

$$\Sigma_c = COP_c^{ec}\eta_{eg}\alpha_{es} + COP_c^{sc}\eta_{st}\varepsilon_{ex}\alpha_{ex} + COP_c^{ac}\varepsilon_{co}\alpha_{co} \tag{25}$$

Similarly, from Eqs. (2), (7) and (9):

$$\frac{\dot{H}_{pb}}{\dot{E}_{load}} = \frac{R_{HE}}{\eta_{pb}} - \frac{\dot{Q}_{recovery}}{\eta_{pb}\dot{E}_{load}} \tag{26}$$

Eqs. (8), (17), (19), (21) and (24) provide:

$$\frac{\dot{Q}_{recovery}}{\dot{E}_{load}} = \frac{\Sigma_h}{\Sigma_c}(R_{CE} + COP_c^{ec}) - COP_h^{ec} \tag{27}$$

where an overall heat pump heating efficiency factor, Σ_h, is equally defined:

$$\Sigma_h = COP_h^{ec}\eta_{eg}\alpha_{es} + COP_h^{sc}\eta_{st}\varepsilon_{ex}\alpha_{ex} + COP_h^{ac}\varepsilon_{co}\alpha_{co} \tag{28}$$

Taking Eqs. (24)–(28) into Eq. (6), a final expression for *ECR* is obtained, in terms of the load ratios, R_{HE} and R_{CE}, and the characteristic parameters of the system components:

$$ECR = \frac{R_{CE} + R_{HE} + 1}{((R_{CE} + COP_c^{ec})/\Sigma_c) + (R_{HE}/\eta_{pb}) - (\Sigma_h/\eta_{pb}\Sigma_c)(R_{CE} + COP_c^{ec}) + (COP_h^{ec}/\eta_{pb})} \tag{29}$$

The amount of fuel burnt to meet all three loads (electricity, heating and refrigeration) can be represented by the energy rate equivalent of fuel consumption to electricity load ratio, R_{FE}, defined as:

$$R_{FE} = \frac{\dot{H}_{total}}{\dot{E}_{load}} \tag{30}$$

From Eqs. (4), (24), (26) and (27), one has:

$$R_{FE} = \left(1 - \frac{\Sigma_h}{\eta_{pb}}\right)\left(\frac{R_{CE} + COP_c^{ec}}{\Sigma_c}\right) + \frac{1}{\eta_{pb}}(R_{HE} + COP_h^{ec}) \tag{31}$$

2.3. Heat demand less than heat recovered

If the heat load is less than or equal to the total heat recoverable, the peak boiler can be dispensed with (this corresponds to $\dot{Q}_{pb} = 0$ and, consequently, $\dot{H}_{pb} = 0$). Then:

$$\dot{Q}_{heating} \le \dot{Q}_{recovery} \tag{32}$$

From Eq. (27), the range of R_{HE} where the heating demand, $\dot{Q}_{heating}$, is less than the heat recovered is given by:

$$R_{HE} \le R_{HE}^* : R_{HE}^* = \frac{\Sigma_h}{\Sigma_c}(R_{CE} + COP_c^{ec}) - COP_h^{ec} \tag{33}$$

From Eqs. (1)–(3) and (29):

$$ECR(R_{HE} \le R_{HE}^*) = \Sigma_c\frac{R_{CE} + R_{HE} + 1}{R_{CE} + COP_c^{ec}} \tag{34}$$

And, from (24):

$$R_{FE}(R_{HE} \le R_{HE}^*) = \frac{R_{CE} + COP_c^{ec}}{\Sigma_c} \tag{35}$$

2.4. Minimum cooling load

For a given electric load, in a power-matched system, a certain amount of cooling effect will be present from the steam driven and absorption chillers, for they are driven by the heat engine waste heat (exhaust and coolant, respectively). If the demand for cooling power is greater than the cooling power produced by the waste heat-driven chillers systems (steam turbine and absorption), then the electrically driven vapor compression chiller comes into play. This lower bound for the cooling load is defined by R_{CE}^*, as follows:

$$R_{CE}^* = \frac{\dot{Q}_{ev}^{sc} + \dot{Q}_{ev}^{ac}}{\dot{E}_{load}} \tag{36}$$

In this situation, all electricity produced goes to the electrical power load, so that, from Eqs. (18) and (20), making the compressor power equal to zero, Eq. (15):

$$R_{CE}^* = \frac{COP_c^{sc}\eta_{st}\varepsilon_{ex}\alpha_{ex} + COP_c^{ac}\varepsilon_{co}\alpha_{co}}{\eta_{eg}\alpha_{es}} \tag{37}$$

2.5. Results

Eqs. (29) and (34) were applied for a set of typical data ($\alpha_{ex} = \alpha_{co} = \alpha_{es} = 0.3$, $COP_c^{sc} = COP_c^{ec} = 3$, $COP_c^{ac} = 0.8$, $COP_h^{sc} = COP_h^{ec} = 4.2$, $COP_h^{ac} = 1.7$, $\eta_{eg} = 0.9$, $\eta_{st} = 0.3$, $\eta_{pb} = 0.9$, $\varepsilon_{ex} = 0.7$, $\varepsilon_{co} = 0.8$). Fig. 2 depicts the variation of the energy conversion ratio as a function of R_{HE} and R_{CE}. It can be seen that, for a given cooling-to-electricity load ratio, there is a certain value for the heating-to-electricity load ratio for which an optimum value for

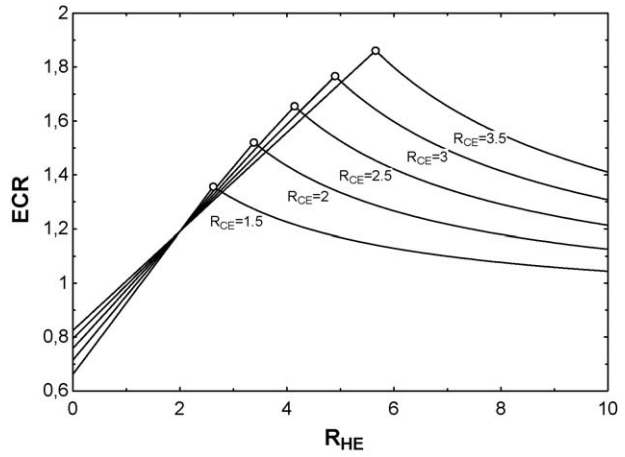

Fig. 2. Variation of the energy conversion ratio with the cooling-to-electricity and heating-to-electricity load ratios for a generic tri-generation system.

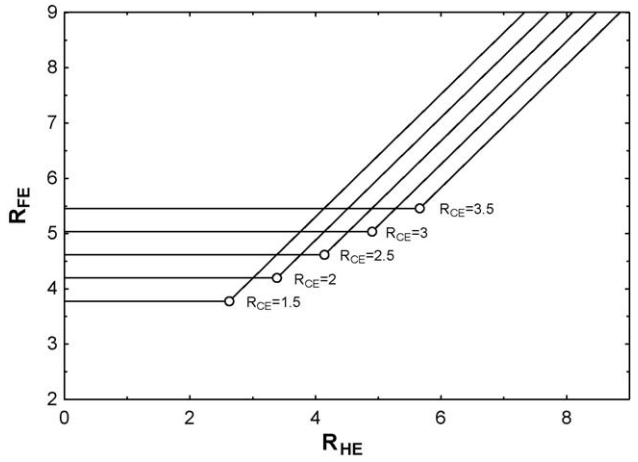

Fig. 3. Variation of the energy rate equivalent of fuel consumption to electricity load ratio with the cooling-to-electricity and heating-to-electricity load ratios for a generic tri-generation system.

ECR is obtained. It corresponds to the situation where the heat load is exactly the recovered heat from the chillers ($R_{HE} = R^*_{HE}$). As explained before, for greater values of R_{HE}, a less efficient heating solution (the peak boiler) is necessary. For lower values, excess waste heat is produced from the fuel, with no use for it. For greater values of R_{CE}, the heat engine is set to operate at greater generator loads (to cope with the chiller compressor power) and a large amount of recovered waste heat is available, thus eclipsing the less efficient effect of the peak boiler operation.

Fig. 3 displays the variation of R_{FE}, related to fuel consumption, with cooling and heating loads, R_{CE} and R_{HE}. For the same reasons described in the preceding paragraph, R_{FE} remains constant for values of R_{HE} below R^*_{HE}. Above it, R_{FE} increases linearly, with Eq. (35).

3. Tri-generation in an office building

3.1. System description

Cardona and Piacentino [3–5] built and studied a Combined Heating and Cooling Power (CHCP) pilot plant, whose operation during both summer and winter corresponds to tri-generation, designed to be installed in Mediterranean areas where climate conditions lead to nearly equal thermal and cooling demands during the year. Their study was aimed at detecting annual trends of energetic demands in full-scale office buildings and to pinpoint the best energy management strategy. To optimize the use of energy, different plant configurations were also examined. The pilot plant which supplies electricity, heating and cooling to two buildings is schematically represented in Fig. 4. The scheme, which is a simplified transcription of their complex CHCP plant, comprises an internal combustion gas-fired four-stroke engine, two (high and low temperature) heat exchangers, one boiler, one pressurized auxiliary boiler, one building heat exchanger, one absorption chiller and one reversible vapor compression heat pump. Some components of the original study (cooling towers, emergency radiator), playing no role in normal summer mode, were excluded from the diagram. In addition, to facilitate the modeling of the (two-building tri-generation) system, details (valves, primary and secondary circuitries of heat exchangers, mufflers, etc.) of the piping network were omitted and replaced by energy flows. Peak values of energy demands (for summer mode) and equipment characteristics are compiled in Tables 1 and 2. Some relevant experimental data were also provided by one of the authors [21]. Recovered waste heat was distributed to either the building heat exchanger or to the absorption chiller, depending on summer ($x < 1$) or winter ($x = 1$) operation mode.

Table 1
Thermodynamic parameters for the two-building tri-generation plant.

Fraction of energy that goes to engine coolant α_{ec}	0.3
Fraction of energy that goes to engine exhaust α_{ex}	0.3
Fraction of energy that goes to engine shaft power α_{shaft}	0.3
Electric generator efficiency η_{EG}	0.9
Peak boiler efficiency η_{PB}	0.9
Coefficient of performance of vapor compressor chiller	2.5
Coefficient of performance of absorption chiller	0.68
Heat recovery efficiency ε	0.86–0.91

3.2. Thermodynamic model

From the definition of *ECR*, similar to the preceding case, Eq. (6):

$$ECR = \frac{R_{CE} + R_{HE} + 1}{(\dot{H}_{he} + \dot{H}_{pb})/\dot{E}_{load}}$$

Here, as opposed to the generic tri-generation system studied above, the total heat recovered, $\dot{Q}_{recovery}$, does not include the heat rejected by the chiller condensers. The contributions are limited to the heat recovered from the engine exhaust and coolant heat exchangers. That is

$$\dot{Q}_{recovery} = (\alpha_{ex}\varepsilon_{ex} + \alpha_{co}\varepsilon_{co})\dot{H}_{he} = \Gamma_{he}\dot{H}_{he} \quad (38)$$

where

$$\Gamma_{he} = (\alpha_{ex}\varepsilon_{ex} + \alpha_{co}\varepsilon_{co}) \quad (39)$$

To compute the total heat load, $\dot{Q}_{heating}$, delivered to the buildings, one must take into account the heat from the pressurized auxiliary boiler. Thus, with Eq. (38):

$$\dot{Q}_{heating} = \varepsilon_{bx}x(\Gamma_{he}\dot{H}_{he} + \eta_{pb}\varepsilon_{bo}\dot{H}_{pb}) \quad (40)$$

where x is the fraction of the total heat available which goes to the building heat exchanger.

It is also noted that:

$$\dot{Q}_{cooling} = \dot{Q}^{ac}_{ev} + \dot{Q}^{ec}_{ev} \quad (41)$$

where

$$\dot{Q}^{ac}_{ev} = COP^{ac}_{c}(1 - x)(\dot{Q}_{recovery} + \eta_{pb}\varepsilon_{bo}\dot{H}_{pb}) \quad (42)$$

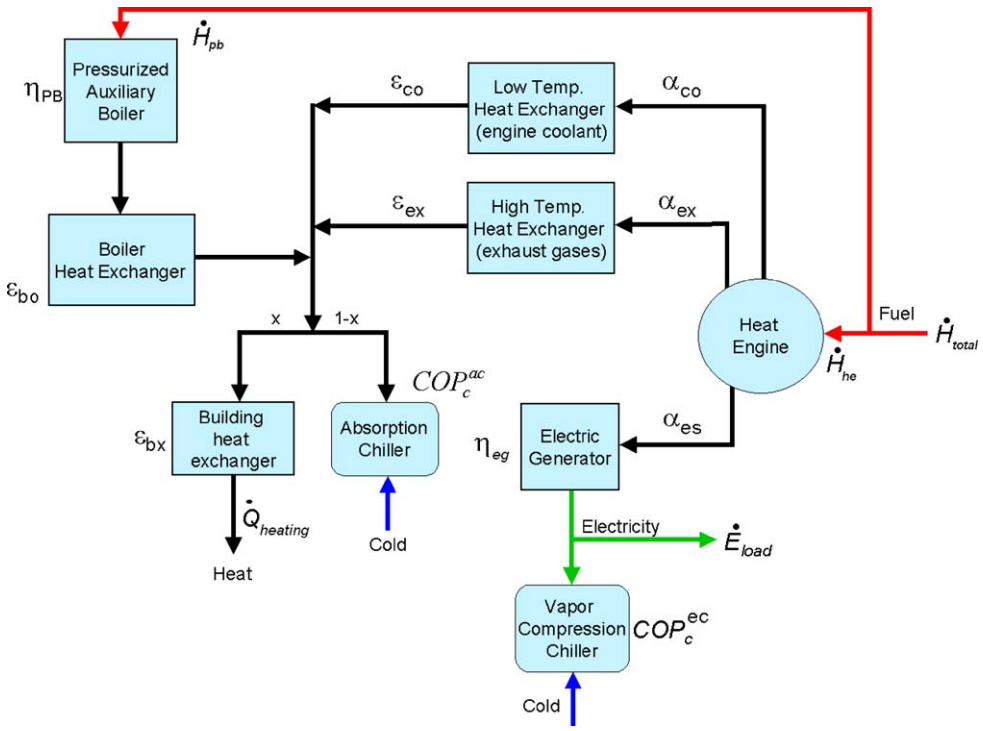

Fig. 4. Schematic diagram of a tri-generation system installed in an office building.

Table 2
Peak values of fuel consumption, engine thermal output and energy demands [4].

	Thermal input (fuel consumption)	Heat engine thermal output	Electric demand	Heating demand	Cooling demand
Summer mode	1450 kW	567 kW	420 kW	70 kW	745 kW

and

$$\dot{Q}_{ev}^{ec} = COP_c^{ec}(\alpha_{es}\eta_{eg}\dot{H}_{he} - \dot{E}_{load}) \tag{43}$$

Eqs. (38) and (41)–(43) provide:

$$\dot{Q}_{cooling} = COP_c^{ac}(1-x)(\Gamma_{he}\dot{H}_{he} + \eta_{pb}\varepsilon_{bo}\dot{H}_{pb}) + COP_c^{ec}(\alpha_{es}\eta_{eg}\dot{H}_{he} - \dot{E}_{load}) \tag{44}$$

And, consequently

$$\dot{H}_{he} = \frac{\dot{Q}_{cooling} - COP_c^{ac}(1-x)(\dot{Q}_{heating}/\varepsilon_{bx}x) + COP_c^{ec}\dot{E}_{load}}{COP_c^{ec}\alpha_{es}\eta_{eg}} \tag{45}$$

From Eq. (40):

$$[\dot{H}_{he} + \dot{H}_{pb}] = \frac{(\dot{Q}_{heating}/\varepsilon_{bx}x) + (\eta_{pb}\varepsilon_{bo} - \Gamma_{he})\dot{H}_{he}}{\eta_{pb}\varepsilon_{bo}} \tag{46}$$

Taking Eq. (45) into Eq. (46):

$$[\dot{H}_{he} + \dot{H}_{pb}] = \frac{\dot{Q}_{heating}}{\eta_{pb}\varepsilon_{bo}\varepsilon_{bx}x} + \frac{\eta_{pb}\varepsilon_{bo} - \Gamma_{he}}{\eta_{pb}\varepsilon_{bo}COP_c^{ec}\alpha_{es}\eta_{eg}} \times \left(\dot{Q}_{cooling} - COP_c^{ac}(1-x)\frac{\dot{Q}_{heating}}{\varepsilon_{bx}x} + COP_c^{ec}\dot{E}_{load}\right) \tag{47}$$

Dividing (47) by $\dot{E}_{load}$ and taking it into Eq. (6), one has the energy conversion ratio for this case:

$$ECR = \frac{R_{CE} + R_{HE} + 1}{(R_{HE}/(\eta_{pb}\varepsilon_{bo}\varepsilon_{bx}x)) + ((\eta_{pb}\varepsilon_{bo} - \Gamma_{he})/(\eta_{pb}\varepsilon_{bo}COP_c^{ec}\alpha_{es}\eta_{eg}))[R_{CE} - COP_c^{ac}(1-x)(R_{HE}/\varepsilon_{bx}x) + COP_c^{ec}]} \tag{48}$$

And, from Eqs. (30) and (47):

$$R_{FE} = \frac{R_{HE}}{\eta_{pb}\varepsilon_{bo}\varepsilon_{bx}x} + \frac{\eta_{pb}\varepsilon_{bo} - \Gamma_{he}}{\eta_{pb}\varepsilon_{bo}COP_c^{ec}\alpha_{es}\eta_{eg}} \times \left(R_{CE} - COP_c^{ac}(1-x)\frac{R_{HE}}{\varepsilon_{bx}x} + COP_c^{ec}\right) \tag{49}$$

3.3. Recovered heat sufficient to meet heat demand

If the heating load demand is lower than the heat recovered from engine exhaust gases and coolant (the portion diverted to the building heat exchanger), then the pressurized auxiliary boiler is not necessary and $\dot{H}_{pb} = 0$. In the limiting case, $\dot{Q}_{heating}^{*}$, when the heat recovered is equal to the heat demand, one has:

$$\dot{Q}_{heating}^{*} = x\dot{Q}_{recovery} \tag{50}$$

From Eqs. (38) and (45):

$$\frac{\dot{Q}_{heating}^{*}}{x\Gamma_{he}} = \frac{\dot{Q}_{cooling} - COP_c^{ac}(1-x)(\dot{Q}_{heating}^{*}/\varepsilon_{bx}x) + COP_c^{ec}\dot{E}_{load}}{COP_c^{ec}\alpha_{es}\eta_{eg}} \tag{51}$$

After rearranging and dividing by $\dot{E}_{load}$, one obtains the expression for the limit value of the cooling-to-electricity load ratio:

$$R_{HE}^{*} = \frac{x(R_{CE} + COP_c^{ec})}{(COP_c^{ec}\alpha_{es}\eta_{eg}/\Gamma_{he}) + (COP_c^{ac}(1-x)/\varepsilon_{bx})} \tag{52}$$

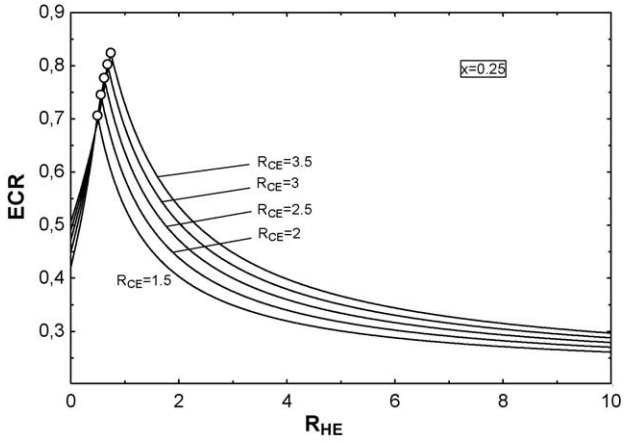

Fig. 5. Variation of the energy conversion ratio with the cooling-to-electricity and heating-to-electricity load ratios for an office-building tri-generation system, $x = 0.25$.

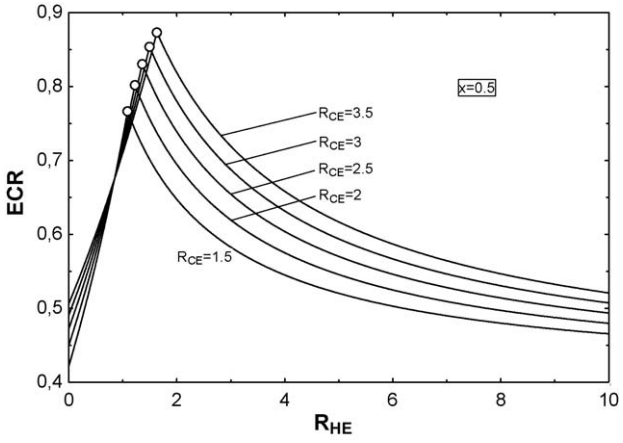

Fig. 6. Variation of the energy conversion ratio with the cooling-to-electricity and heating-to-electricity load ratios for an office-building tri-generation system, $x = 0.5$.

From Eq. (6):

$$ECR(R_{HE} \le R^*_{HE}) = \frac{R_{CE} + R_{HE} + 1}{\dot{H}_{he}/\dot{E}_{load}} \tag{53}$$

and taking Eq. (45) into Eq. (53):

$$ECR(R_{HE} \le R^*_{HE}) = \frac{COP^{ec}_c \alpha_{es} \eta_{eg}(R_{CE} + R_{HE} + 1)}{R_{CE} - ((COP^{ac}_c(1-x))/\varepsilon_{bx}x)R_{HE} + COP^{ec}_c} \tag{54}$$

For this condition, the energy rate equivalent of fuel consumption to electricity load ratio, R_{FE}, is:

$$R_{FE}(R_{HE} \le R^*_{HE}) = \frac{\dot{H}_{he}}{\dot{E}_{load}} \tag{55}$$

From Eq. (45):

$$R_{FE}(R_{HE} \le R^*_{HE}) = \frac{R_{CE} - COP^{ac}_c(1-x)(R_{HE}/\varepsilon_{bx}x) + COP^{ec}_c}{COP^{ec}_c \alpha_{es} \eta_{eg}} \tag{56}$$

3.4. Results

Cardona and Piacentino [4] investigated several configurations, among which only two were considered in the present work: case D (CHCP with 75% cooling) and case E (CHCP with 50% cooling) where the absorption chillers were, respectively, sized to provide 570 kW and 380 kW of cooling at constant engine power output (475 kW). Inserting the values of Tables 1 and 2 in each case, values of 0.405 and 0.425 were obtained for an energy conversion efficiency for a peak experimental value of 0.85 (the latter being the sum of the energy demands divided by the engine thermal input; Table 2), confirming the results of [4].

Figs. 5–7 show the variation of ECR and R_{FE} with R_{CE} and R_{HE}. The same trend of the preceding case, generic tri-generation, was found. As discussed in the preceding case, maximum energy conversion ratios and minimum fuel consumption are attained when the heating demand equals the available recovered engine waste heat. Fig. 7 summarizes the relation between fuel consumption, R_{FE}, and heating and cooling demands, R_{HE} and R_{CE}, respectively, all the parameters non-dimensionalized in terms of the electrical load demand, $\dot{E}_{load}$. Figs. 8 and 9 show the variation of ECR and R_{FE} with x, for given values of R_{HE}. Large values of R_{HE}, e.g., $\dot{Q}_{heating}$ four to five times greater than $\dot{E}_{load}$, imply lower values of ECR and greater fuel consumption, for increasing x. In this case, better operation is obtained by diverting more of the recovered heat to the building heat exchanger. For lower values of R_{HE}, 3 or less, maximum energy efficiency is attained at a certain value of x, below

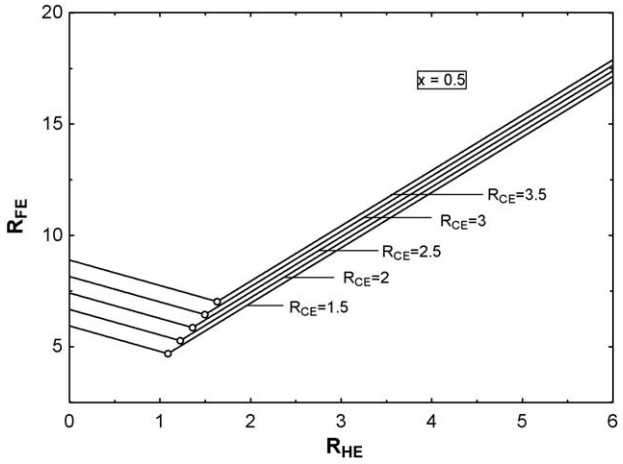

Fig. 7. Variation of the energy rate equivalent of fuel consumption to electricity load ratio with the cooling-to-electricity and heating-to-electricity load ratios for an office-building tri-generation system, $x = 0.5$.

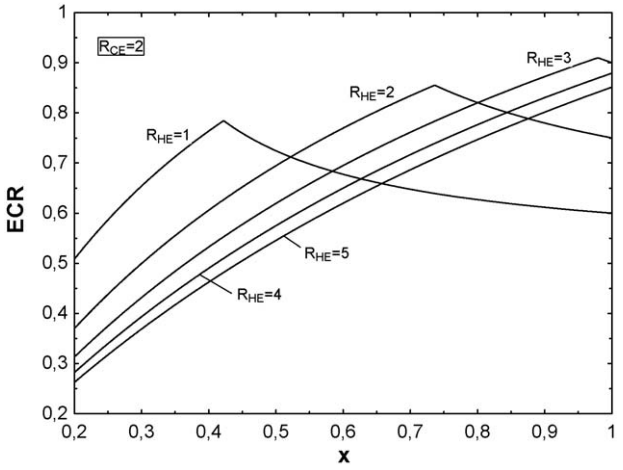

Fig. 8. Variation of the energy conversion ratio with the fraction of the heat recovered from the exhaust and water cooling that goes to the building heat exchanger, for an office-building tri-generation system, $R_{CE} = 2$.

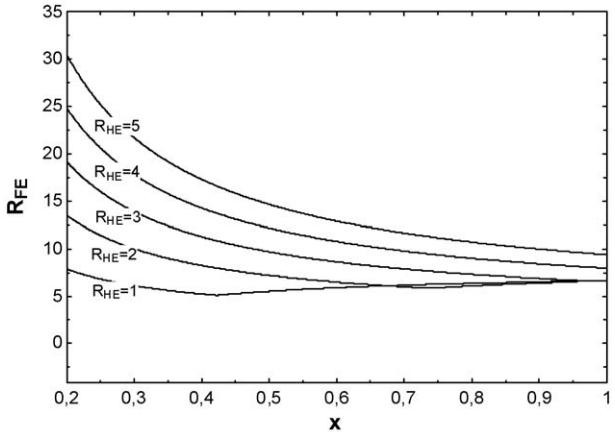

Fig. 9. Variation of the energy rate equivalent of fuel consumption to electricity load ratio with the fraction of the heat recovered from the exhaust and water cooling that goes to the building heat exchanger, for an office-building tri-generation system, $R_{CE} = 2$.

which more heat is required from the less efficient auxiliary boiler. The existence of local optimum values of *ECR* and R_{FE}, for given values of the cooling and heating loads, demonstrate the importance of a well established operation strategy, as the fraction of the heat recovered from the exhaust and water cooling that goes to the building heat exchanger, x, is clearly an operational control variable.

4. Conclusions

Tri-generation may play a significant role in the international effort (on the part of the signatories of the Kyoto Protocol) to reduce CO_2 and other greenhouse gases emissions, since the highly integrated character of tri-generation implies higher energy conversion ratios and hence lower pollutant emissions. Furthermore, tri-generation is sufficiently flexible to permit installation in locations where electricity from the national grid is, for some reason, not available. Given the various conflicting design criteria such as cost, efficiency and environmental impact, plant designers need some sort of "thermodynamic yardstick" to select the best or the most appropriate plant configuration. The first-law analysis presented here for a generic tri-generation system accurately reflects the energy conversion efficiency of the case studied but can, in fact, be applied to any tri-generation system. The introduction of non-dimensional parameters relating cooling and heating demands and fuel energy-equivalent consumption to the electrical load demand, R_{CE}, R_{HE} and R_{FE}, has proven to be an adequate approach that can be extended to second-law or thermo-economic analyses.

The present study is in no way comprehensive, but thermal engineers may find the method of value when sizing and choosing thermal equipment for buildings and other systems.

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